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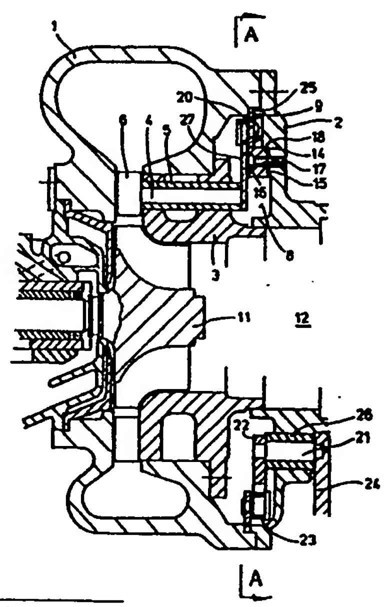
(54) **Variable displacement turbocharger.**

(57) A variable displacement turbocharger comprises a turbine casing (1), a shroud (3) coaxially within the casing and a gas outlet cover (2) connected to the casing. The casing (1), the shroud (3) and the gas outlet cover (2) together define a closed annular space (8). The casing (1) and the shroud (3) together define a space which accommodates a turbine wheel (11) and which communicates with a gas inlet passage within the casing (1) through an annular passage in which a plurality of nozzle blades (6) are situated. Each nozzle blade (6) is rotatably carried by a nozzle shaft (4) and the nozzle shafts (4) are connected by respective links (27) to a common nozzle driving ring (9) within the annular space (8). The driving ring (9) is connected to actuating means (21,24) outside the space (8), movement of which results in rotation of the driving ring (9) and thus of the nozzle blades (6) to adjust the flow rate of exhaust gases to the turbine wheel. The driving ring (9) is axially retained by the accommodation of its periphery in a space defined between the casing (1) and the gas outlet cover (2).

Fig. 1



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Description

VARIABLE DISPLACEMENT TURBOCHARGER

The present invention relates to turbochargers and is concerned with variable displacement turbochargers of the type comprising a turbine casing, a shroud within the casing and a gas outlet cover connected to the casing, the casing, the shroud and the gas outlet cover together defining a closed annular space and the casing and the shroud together defining a space which accommodates a turbine wheel and which communicates with the gas inlet passage within the casing through an annular passage in which a plurality of nozzle blades are situated, each nozzle blade being rotatably carried by a nozzle shaft and the nozzle shafts being connected by respective links to a nozzle driving ring within the annular space, the driving ring being connected to actuating means outside the space, movement of which results in rotation of the driving ring and thus of the nozzle blades.

Turbochargers, which serve to improve the thermal efficiency of various types of prime movers or engines, are driven by the exhaust gases from the engines. The flow rate of exhaust gases from an engine varies depending on the load on the engine so that conventionally part of the performance of the turbocharger is sacrificed. To overcome this problem, variable turbochargers have recently been developed in which the angle of the nozzle blades is varied depending on the load on the engine so as to control the flow rate of exhaust gases flowing to the turbine wheel in an optimum manner, whereby a high degree of thermal efficiency can be maintained at all times.

The turbine unit of a known variable displacement turbocharger of the type described above is shown in Figures 11 and 12 in which Figure 11 is a side sectional view and Figure 12 is a transverse sectional view on the line C-C in Figure 11. The turbine unit includes a shroud 3 which is clamped between a turbine casing 1 and a gas outlet cover 2 and rotatably carries nozzle shafts 4 in bearings 5. A nozzle blade 6 is securely attached to one end of each nozzle shaft 4 on the turbine casing side and one end of a nozzle link 7 is attached to the other end of the nozzle shaft 4 on the gas outlet cover side.

An annular or generally toroidal space 8 defined between the gas outlet cover 2 and the shroud 3 accommodates the nozzle links 7 and a nozzle driving ring 9 positioned and rotatably supported by projections of the bearings 5. The nozzle driving ring 9 is connected by intermediate links 10 to the other ends of the nozzle links 7. A turbine wheel 11 is mounted within the space defined by the shroud 3 and casing 1.

Rotation of the nozzle driving ring 9 by an external actuator connected to a lever 13 causes the links 7 and 10 to move and thus the nozzles blades 6 to rotate by an angle corresponding to the angle of rotation of the nozzle driving ring 9.

The nozzle blades 6 are situated in the constricted annular passage between the interior of the casing 1 and the space accommodating the turbine wheel. When the nozzle driving ring 9 is rotated in response to the load on the engine, the angle of the nozzle blades 6 is changed to optimise the flow rate and the angle of entry of the exhaust gases flowing against the turbine wheel 11, whereby the efficiency of the turbocharger is improved.

In the turbocharger of the type described above, the nozzle driving ring 9 is axially restrained by the nozzle links 7 and the intermediate links 10. This restraint is not particularly positive due to play between the bearings 5 and nozzle shafts 4, between the intermediate links 10 and nozzle driving ring 9 and between the nozzle links 7 and the intermediate links 10. There is thus a risk that vibration of the turbocharger may occur during the operation. The axial vibratory load of the relatively heavy nozzle driving ring 9 is received by the links, which may cause damage to the connections between the links and wear of the sliding component parts.

High temperature exhaust gases flow through the turbine casing 1 so that it and the shroud 3 which is in intimate contact therewith and is partly exposed to contact with the exhaust gases become extremely hot. As a result, the temperature of the bearings 5 in the shroud 3 also rises. The nozzle driving ring 9, which is supported by the bearings 5 as described above, is not directly exposed to the exhaust gases and its temperature rises later than that of the turbine casing 1 and the shroud 3. As a result, the positions of the bearings 5 may vary in the radial direction within the nozzle driving ring 9 which has not yet thermally expanded so that the clearance between the inner surface of the nozzle driving ring 9 and the outer peripheral surfaces of the bearing 5 disappears, thereby resulting in contact and sticking between the nozzle driving ring 9 and the bearings 5.

In the known turbocharger, the weight of the nozzle driving ring 9 is received by only a few of the bearings 5 at their upper extremities through a line contact between the nozzle driving ring 9 and these bearings so that the pressure loading on these bearings 5 is very considerable. Therefore when the nozzle driving ring 9 is rotated in sliding contact with these bearings a progressive wear occurs. If the sliding contact surfaces of the bearings 5 are worn, the nozzle driving ring 9 becomes eccentrically disposed so that the angles of the nozzle blades which depend on the angular position of the nozzle driving ring, become non-uniform. As a result, the turbocharger cannot exhibit the desired optimum performance.

Furthermore, when the opening angle, i.e. the degree of openness, of the nozzle blades is too great, the nozzle blades may contact the turbine wheel and damage it. It follows therefore that control of the opening angle of the nozzles blades is necessary to prevent such contact. In the turbocharger described above, the opening angle is adjusted only on the drive or actuation side of the

nozzle driving ring and no adjustment is made on the turbine side. Therefore when the angle control on the drive side of the nozzle driving ring cannot be performed properly, the nozzle blades are caused to open so widely that they contact the turbine blade. causing damage to both the nozzle blades and the turbine wheel.

The smaller the gaps between the nozzle blades 6, the turbine casing 1 and the shroud 3, the higher is the guiding effect on the exhaust gases by the nozzle blades 6 and thus the efficiency of the turbine. In particular, a gap at the root of the nozzle blade 6 (i.e. between the nozzle blade 6 and the shroud 3) has a greater influence on the guiding effect than does a gap at the tip of the nozzle blade 6 (i.e. between the nozzle blade 6 and the turbine casing 1). It follows that the gap at the root of the nozzle blades 6 must be reduced as much as possible but the extent to which this is possible is limited, as will be described below.

The size of the gap varies with thermal deformation of the shroud and the nozzle blades. Such variation is also caused by the oxidation-roughened surfaces of the shroud and nozzle blades and by misalignment of the nozzle shaft 4 due to wear of the bearings 5. Thus if the root gap is too small, the nozzle blade 6 may contact the shroud 3 and impede movement of the nozzle blade 6 or cause sticking between the nozzle blade 6 and shroud 3.

In the construction described above, the nozzle shaft 4 has a throat 40 at its end adjacent the nozzle blade 6 whose diameter is greater than that of the remainder of the nozzle shaft 4 and is substantially equal to the outer diameter of the bearing 5. The bearing 5 is fitted into the shroud 3 to abut the throat 40 with the flange surface of the throat 40 contacting the end surface of the bearing 5.

With this structure, the nozzle blades 6 are generally urged by the pressure of the exhaust gases toward the shroud 3 so that there is considerable friction between the nozzle shaft 4 and the bearing 5. This significantly influences the capacity of the actuator to move the nozzle blades. It becomes difficult to correctly control the angle of the nozzle blades since such high friction will cause hysterisis of the movement of the nozzle blades.

In order to ensure sufficiently smooth movement of the moving parts, there must be an adequate gap between the nozzle shafts 4 and the bearings 5. However, this is disadvantageous in that the exhaust gases flow through the space between the nozzle shafts 4 and the bearings 5 into the space 8 and then leak to the atmosphere through the space between a shaft 4 of the lever 13 and the associated bush. Such leakage of the exhaust gases contaminates the surrounding atmosphere and is dangerous because of the high temperature of the exhaust gases.

It is an object of the invention to overcome the above disadvantages and to provide a turbocharger of the type referred to in which the nozzle driving ring is effectively restrained in the axial direction with a high degree of accuracy. A further object of the present invention is to ensure that smooth rotation of the nozzle driving ring can occur at all times without sticking, thereby enabling variation of the angle of the nozzle blades easily and accurately. A further object is to permit the control of the maximum opening angle of the nozzle blades, to prevent contact between the nozzle blades and the turbine wheel. A further object is to avoid contact between the nozzle blades and the shroud and to decrease the friction between the nozzle blades and the bearings. A final object of the present invention is to prevent leakage of exhaust gases from the mechanism for adjusting the angle of the nozzle blades.

According to the present invention a turbocharger of the type referred to above is characterised in that the periphery of the driving ring is slidably received in a space defined between the casing and the gas outlet cover. This feature ensures that the driving ring is reliably axially retained but may nevertheless rotate freely. In the preferred embodiment the nozzle driving ring is rotatably supported by one or more projections on the gas outlet cover, which projections are preferably in sliding contact with the internal surface of the driving ring. The projection or projections may be integral with the gas outlet cover and in this event may constitute either a single continuous circular ring or a plurality of spaced projections. The external sliding contact surface of the projection or projections may be hardened to minimise wear. Alterntively the projections may constitute separate items connected to the gas outlet cover and in one embodiment there is a plurality of projections, each of which comprise a centering fixture removably connected to the gas outlet cover by means of a countersunk-head screw.

It is preferred that the radius of curvature of at least the end portions of the sliding contact surface of the or each projection is smaller than that of the inner surface of the driving ring. The projections need not be all of the same size nor symmetrically disposed and both the size and disposition of the projections may be such as to produce a uniform contact pressure on all of them. In one embodiment. certain of the projections have larger surfaces than the remainder so as to accommodate the load transmitted from the nozzle driving ring.

The shroud may have one or more protuberances or projections, preferably continuous annular projections, positioned to be engaged by the nozzle blades when they reach a predetermined position. This construction ensures that the nozzle blades cannot come into contact with the turbine wheel. In an alternative embodiment at least one of the centering projections is positioned to be engaged by the nozzle driving ring or a member connected thereto when the nozzle blades have reached a predetermined position.

In one embodiment the nozzle shafts have a uniform diameter over their length and are supported in bearings having an end face which directly contacts the associated nozzle blade. Alternatively or in addition, the side surface of the nozzle blade adjacent the shroud may be inclined away from the shroud to eliminate the risk of contact between the nozzle blades and the shroud.

The leakage of exhaust gas to the atmosphere around the actuating means may be eliminated by

supplying compressed air into the annular space through a gas connection or alternatively by providing gas communication between the annular space and the gas outlet.

Further features and details of the invention will be apparent from the following desription of certain specific embodiments which is given by way of example with reference to Figures 1 to 10 of the accompanying drawings, in which:-

Figure 1 is an axial sectional view illustrating the major components of a variable displacement turbocharger in accordance with the present invention;

Figure 2 is a transverse sectional view on the line A-A in Figure 1;

Figure 3(A) and 3(B) show two examples of integral centering projections;

Figure 4 is a scrap axial view of a turbocharger including a plurality of asymmetrical centering projections;

Figure 5 is a view similar to Figure 1 of an embodiment in which the shroud includes a projection limiting the travel of the nozzle blades;

Figure 6 is a view similar to Figure 2, and illustrating an alternative means for limiting the travel of the nozzle blades;

Figure 7 is a scrap enlarged view of a further embodiment;

Figure 8 is a view similar to Figure 1 of a further embodiment;

Figure 9 is a similar view of a further embodiment incorporating means for preventing the leakage of exhaust gases; and

Figure 10 is a scrap view corresponding to the portion B in Figure 9 illustrating alternative means for preventing the leakage of exhaust gases.

Referring firstly to Figures 1 and 2, the turbocharger includes a turbine casing 1 defining an annular cavity of generally conventional shape, a shroud 3 coaxially disposed within and engaging the casing 1 and a gas outlet cover 2 attached to the turbine casing 1. Within the space defined by the casing 1 and shroud 3 is a turbine wheel 11. The shroud 3 and outlet cover 2 define a gas outlet 12 and the casing 1, shroud 3 and outlet cover 2 define an annular space 8. Circumferentially spaced, axially extending bearings 5 are fitted into the shroud 3 around its periphery. A nozzle shaft 4 is fitted into each bearing 5 and carries at one end a nozzle blade 6 disposed in the restricted passage between the interior of the casing 1 and the space accommodating the turbine wheel. A bifurcated nozzle link 27 is attached to the other end of each nozzle shaft 4 which extends into the space 8. The casing 1 and outlet cover 2 define an annular groove 25 in which the periphery of a nozzle driving ring 9 is slidably accommodated. The outlet cover 2 is formed at an inner surface with a stepped or recessed surface 14 into which centering fixtures 15 are fitted at a predetermined pitch with countersunk-head screws 16. The size of the fixtures 15, the position of the tapped holes 17 for the screws 16 and the position of the step 18 are such that when the centering fixtures are held in position by the screws 16, the centering fixtures 15 are pressed against the step 18 by the tapered surface of the screw heads. If the stepped portion 18 is defined with a high degree of dimensional accuracy, the centering fixtures 15 can be positioned with a high degree of accuracy also.

The inner peripheral surface of the nozzle driving ring 9 slidably contacts the centering fixtures 15 and is thus reliably centered. Thus, the nozzle driving ring 9 is axially and radially positioned by the turbine casing 1, the gas outlet cover 2 and the centering fixtures 15. A dog 20 is rotatably mounted by a pin 19 on the nozzle driving ring 9 and is fitted into the forked leading end of each nozzle link 27.

A shaft 21 passes through a bearing 26 in the gas outlet cover 2 and has a bifurcated driving link 22 securely fixed to its inner end. The forked leading end of the link 22 is engaged with a dog 23 pivotally attached to the nozzle driving ring 9. A lever 24 is attached to the outer end of the shaft 21.

When the lever 24 is pivoted by an external actuator (not shown), the driving link 22 is pivoted by the shaft 21 and rotates the nozzle driving ring 9. This causes pivotal movement of the nozzle links 27 and thus rotation of the nozzle shafts 4, thereby changing the angle of the nozzle blades 6. During this process the nozzle driving ring 9 is restrained radially by the centering fixtures 15 and axially by the groove 25.

The dimensions of the groove 25 and its position can be determined with a high degree of accuracy when the turbine casing and the gas outlet cover are fabricated. The axial clearance between the sides of the groove 25 and the nozzle driving ring 9 can be accurately predetermined and thus the axial restraint of the nozzle driving ring 9 can be maintained very satisfactorily. Thus vibrations due to undesired or excessive play or clearances can be eliminated. Moreover, axial vibratory load of the nozzle driving ring 9 is received not by the links but by the turbine casing 1 and the gas outlet cover 2 so that the turbocharger is highly reliable in operation.

Instead of the centering fixtures 15, each of which is fabricated as a separate component, one or more projections integral with the gas outlet cover may be used. The projections may be in the form of a plurality of column-shaped projections or a continuous circular ring.

There is a possibility that when the nozzle driving ring 9 is driven, the centre of the peripheral or circumferential surface of the ring 9 may deviate from the centre defined by the projection or projections.In order to ensure smooth rotation of the nozzle driving ring 9 even in the above-described off-centre situation, it suffices that the radius of the surface of contact of the projection (especially, the column-shaped projection) is smaller than the radius of the inner peripheral surface of the nozzle driving ring 9.

Figure 3(A) shows one possible configuration of such a projection 15. In this case, the relation $R = \alpha R_o$ is maintained where R is the radius of the surfaceof contact of the projection; R_o is the radius of the inner surface of the nozzle driving ring 9; and α is a coefficient less than 1.

Figure 3(B) shows an alternative configuration in which the projection 15 has a compound surface in which the centre portion of the surface of contact has a radius R_0 equal to that of the inner surface of the nozzle driving ring and the end portions have a smaller radius R_1.

Since the configurations of the projection are determined in the manner described above, the surfaces of the nozzle driving ring and the projection or projections remain in contact and even if eccentric displacement occurs in the rotation of the nozzle driving ring, galling between the nozzle driving ring and the centering fixtures can be prevented to ensure smooth rotation of the nozzle driving ring for an extended period of time.

The configuration of the projections can be improved if the surface pressure exerted on them is taken into consideration. Load is not uniformly distributed over the different projections and may differ in dependence on the weight of the nozzle driving ring 9. In the embodiment of Figure 4, the projections are assymetrically disposed with two projections 15 disposed in the direction of the exerted load being wider than the remaining projections 15'. The surface pressure is thus uniformly distributed without increasing the overall frictional resistance between the projections and the nozzle driving ring resulting in a reduction of wear to a minimum. The wear resistance may be further improved by hardening the sliding surface of the projections.

In the embodiment of Figure 5 an annular projection or protuberance 28 is formed on the shroud 3 adjacent to and downstream of the nozzle blades 6 which extends into the passage from the space within the flow passage 25 at the inlet toward the turbine wheel casing 1 to the turbine wheel. The height h of the projection 28 is greater than the width of the gap g between the nozzle blades 6 and the surface of the shroud 3 so that when the nozzle blades are opened to the maximum opening degree, they contact the projection 28 and are prevented from moving any further. The projection 28 is positioned to ensure that the nozzle blades 6 can not contact the turbine wheel 11.

The projection 28 also serves to guide any exhaust gases which may have leaked through the gap 9 towards the turbine wheel 11 so that the efficiency of guiding the exhaust gases is improved.

The maximum degree of opening of the nozzle blades 6 is controlled in the embodiment of Figure 6 by the provision of a projection 15" adjacent to the nozzle driving link 22 such that it is engaged by the nozzle driving link 22 when the nozzle blades 6 are opened to the desired maximum. Thus when the nozzle blades 6 are opened to the maximum degree, further movement of the link 22 and thus of the nozzle blades 6 is restrained.

As shown most clearly in Figure 7, the bearings 5 which rotatably support the nozzle shafts 4 are of sufficient length that their end face directly contacts the associated nozzle blade 6. This reduces the area of contact and the wear that results on movement of the nozzle blades.

In the embodiment of Figure 8, the bearings 5 terminate flush with .he surface of the shroud 3 in order to prevent contact between the surface of the shroud and the surface of the nozzle blades due to thevariation of the gap therebetween. The side surface of the nozzle blades adjacent to the shroud is inclined in the direction away from the surface of the shroud by an angle β thereby defining a gap.

In the embodiment of Figure 9, the space 8 communicates via a hose 29 with a compressed air source (not shown), whereby the pressure in the space 8 is maintained higher than the pressure of the exhaust gases. The air introduced into the space 8 flows through the gap between the gas outlet cover 2 and the shroud 3 toward the exhaust gas outlet 12 so that the mechanism for angularly moving the nozzle blades is cooled by the air. The gas leaking through the space between the bearing 26 and the shaft 21 is thus low-temperature air.

In the embodiment of which only a portion is illustrated in Figure 10, the space 8 communicates with the gas outlet 12. More particularly, at the joint between the shroud 3 and the gas outlet cover 2, the engaging portion of the shroud is decreased in diameter or the gas outlet cover is increased in diameter, whereby a gap δ is defined therebetween. In use, gas within the space 8 therefore flows into the gas outlet 12 in which exhaust gases flow at a high velocity. These exhaust gases produce an ejection action and positively suck the gases through the gap δ. As a result, no exhaust gases leak through the space between the shaft 21 and the bearing 26 to the exterior.

Alternatively, an axial groove or grooves (not shown) may be formed on the engaging portions of either the shroud or the gas outlet cover or a radial through-hole or holes may extend through the shroud or the gas outlet cover.

Claims

1. A variable displacement turbocharger comprising a turbine casing (1), a shroud (3) within the casing and a gas outlet cover (2) connected to the casing, the casing (1), the shroud (3) and the gas outlet cover (2) together defining a closed annular space (8) and the casing (1) and the shroud (3) together defining a space which accommodates a turbine wheel (11) and which communicates with the gas inlet passage within the casing (1) through an annular passage in which a plurality of nozzle blades (6) are situated, each nozzle blade (6) being rotatably carried by a nozzle shaft (4) and the nozzle shafts (4) being connected by respective links (27) to a nozzle driving ring (9) within the annular space (8), the driving ring (9) being connected to actuating means (21,24) outside the space (8), movement of which results in rotation of the driving ring (9) and thus of the nozzle blades (6), characterised in that the periphery of the driving ring (9) is slidably received in a space defined between the casing (1) and the gas outlet cover (2).

2. A turbocharger as claimed in claim 1 characterised in that the nozzle driving ring (9) is rotatably supported by one or more projections (15) on the gas outlet cover (2).

3. A turbocharger as claimed in claim 2 characterised in that the projections (15) are integral with the gas outlet cover (2).

4. A turbocharger as claimed in claim 2 characterised by a plurality of projections, each of which comprises a centering fixture (15) removably connected to the gas outlet cover (2) by means of a countersunk head screw (17).

5. A turbocharger as claimed in claim 2 or claim 3 characterised in that at least one of the projections (15") is positioned to be engaged by the nozzle driving ring (9) or a member (22) connected thereto when the nozzle blades (6) have reached a predetermined position.

6. A turbocharger as claimed in any one of claims 2 to 5 characterised in that at least the end portions of the sliding contact surface of the or each projection (15) is less than that of the inner peripheral surface of the nozzle driving ring (9).

7. A turbocharger as claimed in any one of claims 2 to 6 characterised by a plurality of projections (15) of which those which, in use, are subjected to a greater proportion of the load transmitted through the nozzle driving ring (9) have a larger surface contacting the nozzle driving ring (9) than the remaining projections (15').

8. A turbocharger as claimed in any one of the preceding claims characterised by one or more protuberances (28) on the shroud positioned to be engaged by the nozzle blades (6) when they reach a predetermined position.

9. A turbocharger as claimed in any one of the preceding claims characterised in that the nozzle shafts (4) have a uniform diameter over their length and are supported in bearings (5) having an end face which directly contacts the associated nozzle blade (6).

10. A turbocharger as claimed in any one of the preceding claims characterised in that the side surface of the nozzle blades (6) adjacent the shroud (3) is inclined away from the shroud (3).

11. A turbocharger as claimed in any one of the preceding claims characterised by a gas connection (29) communicating with the annular space (8) through which, in use, compressed air is introduced into the annular space (8)

12. A turbocharger as claimed in any one of claims 1 to 10 characterised in that the annular space (8) and the gas outlet (12) are in communication.

Fig. 1

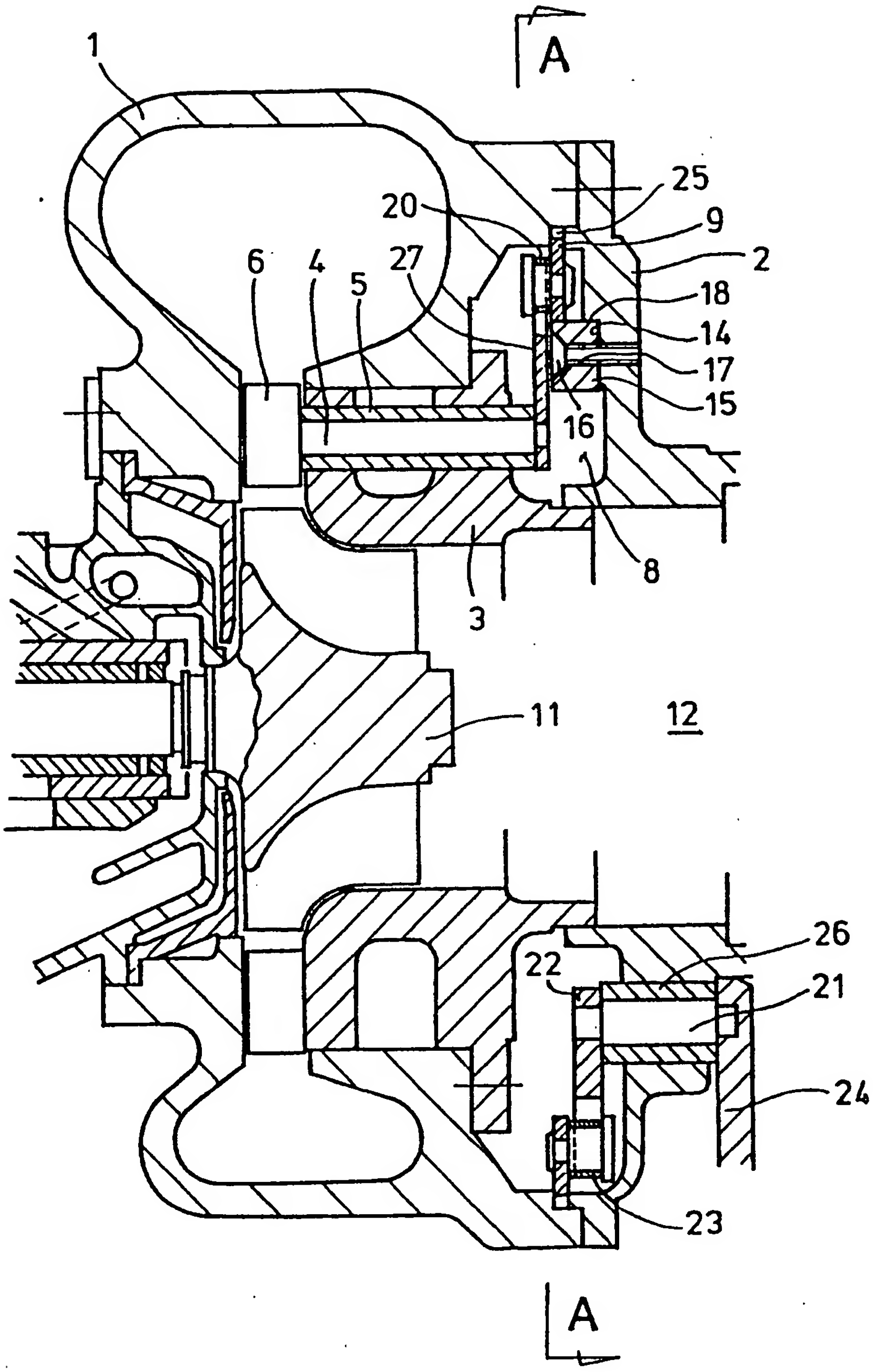

Fig.2

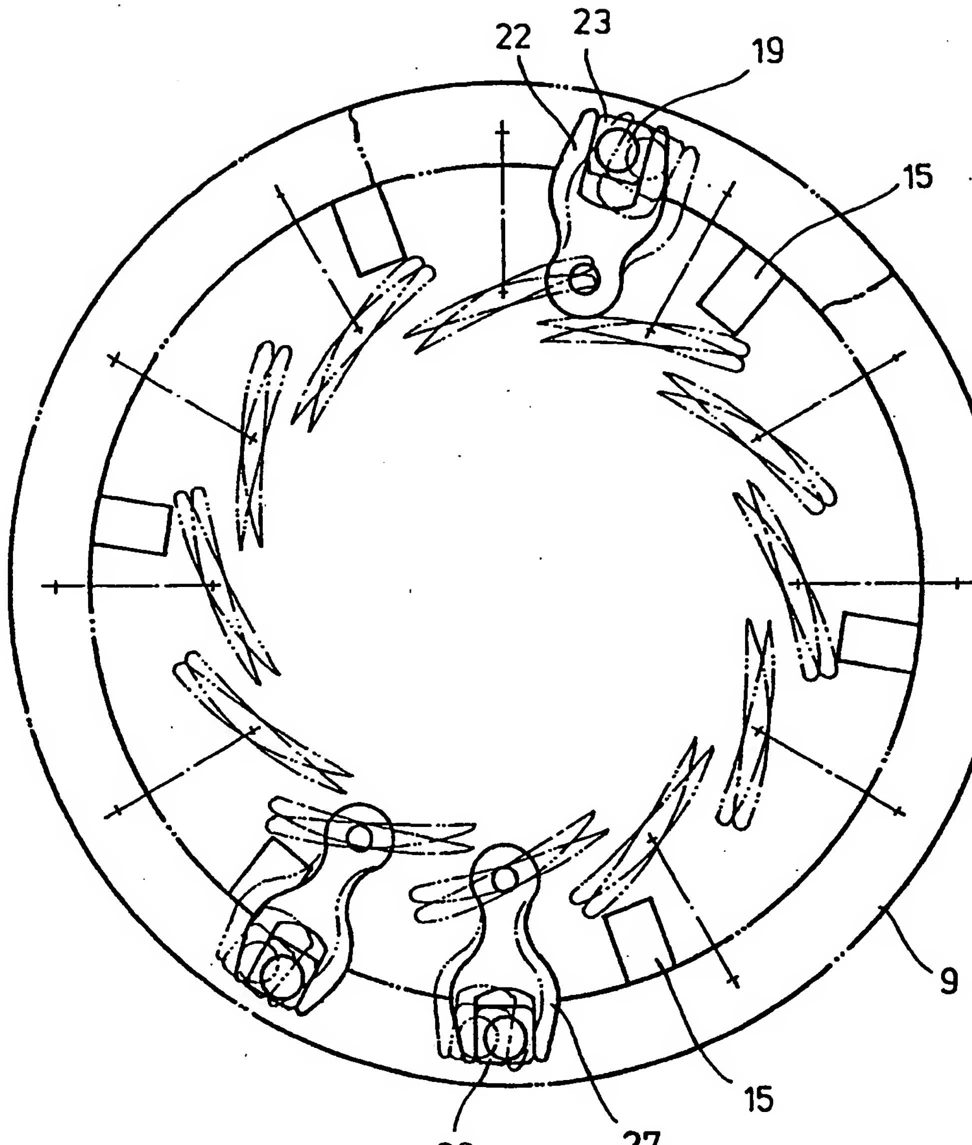

Fig.3(A)

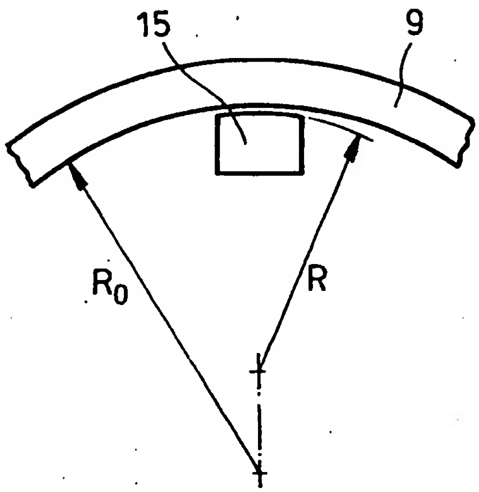

Fig.3(B)

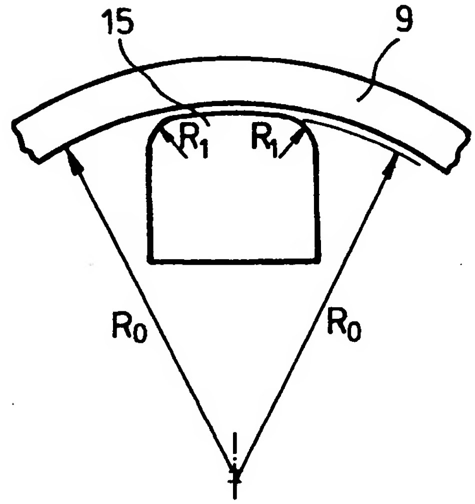

Fig.7

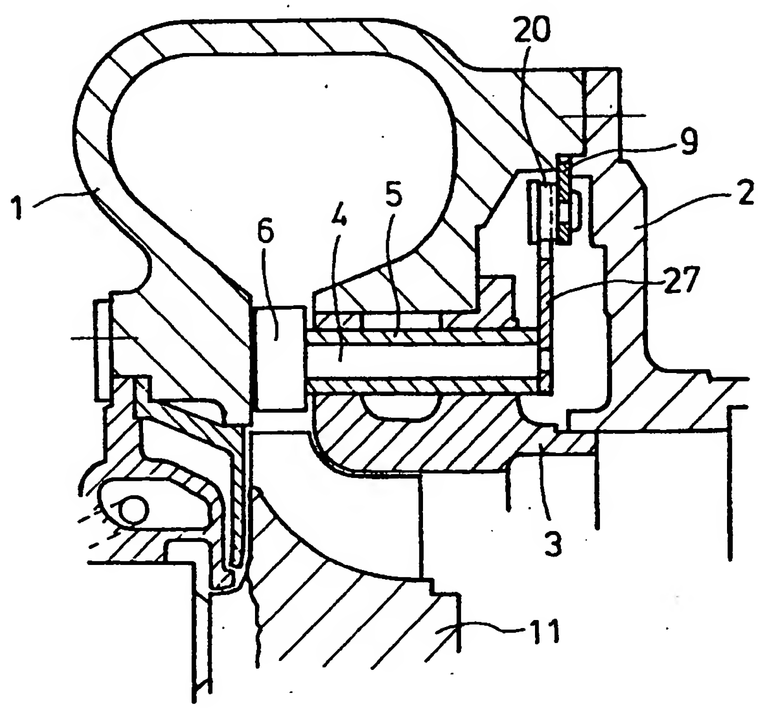

Fig.4

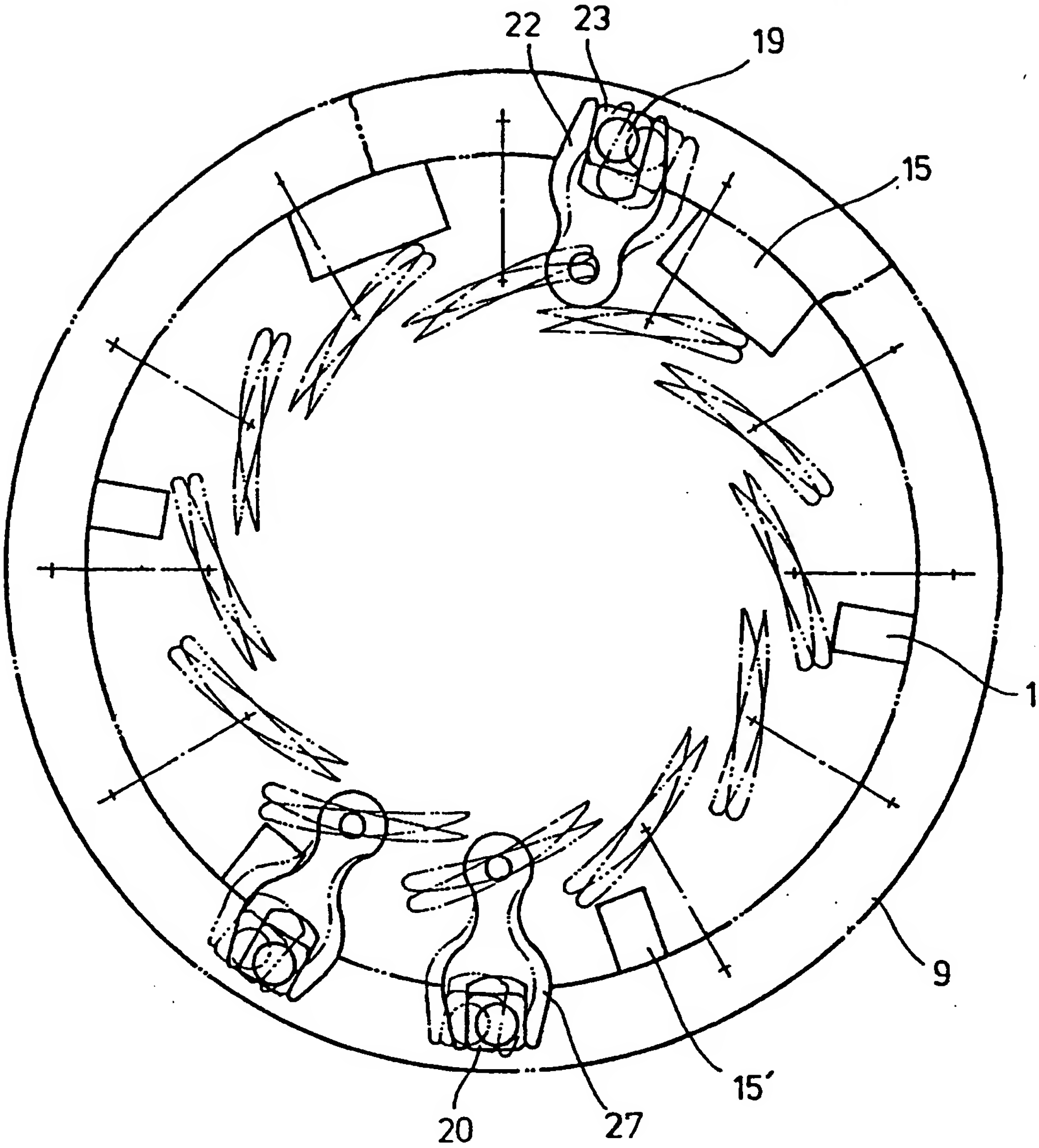

Fig.5

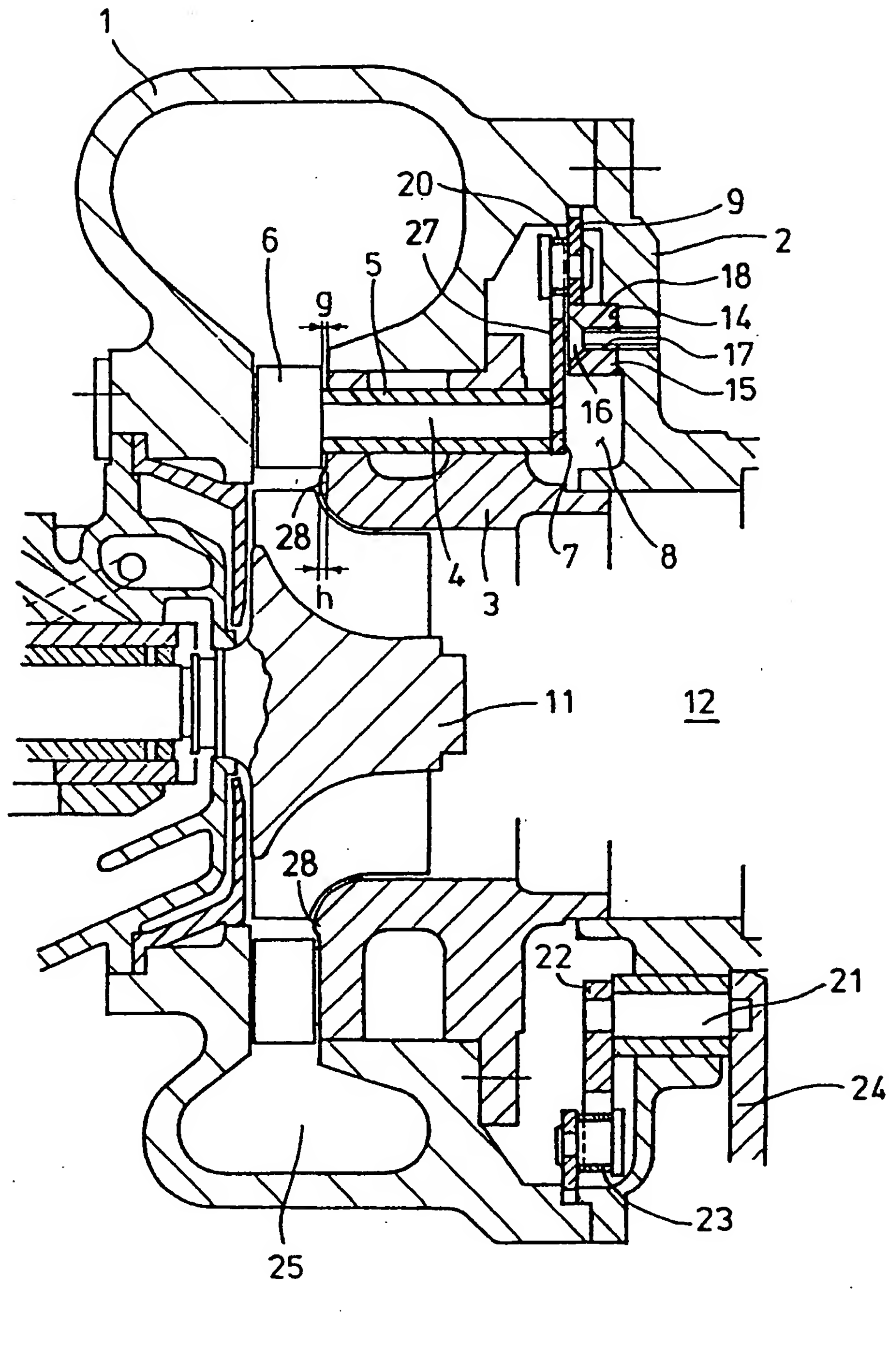

Fig.6

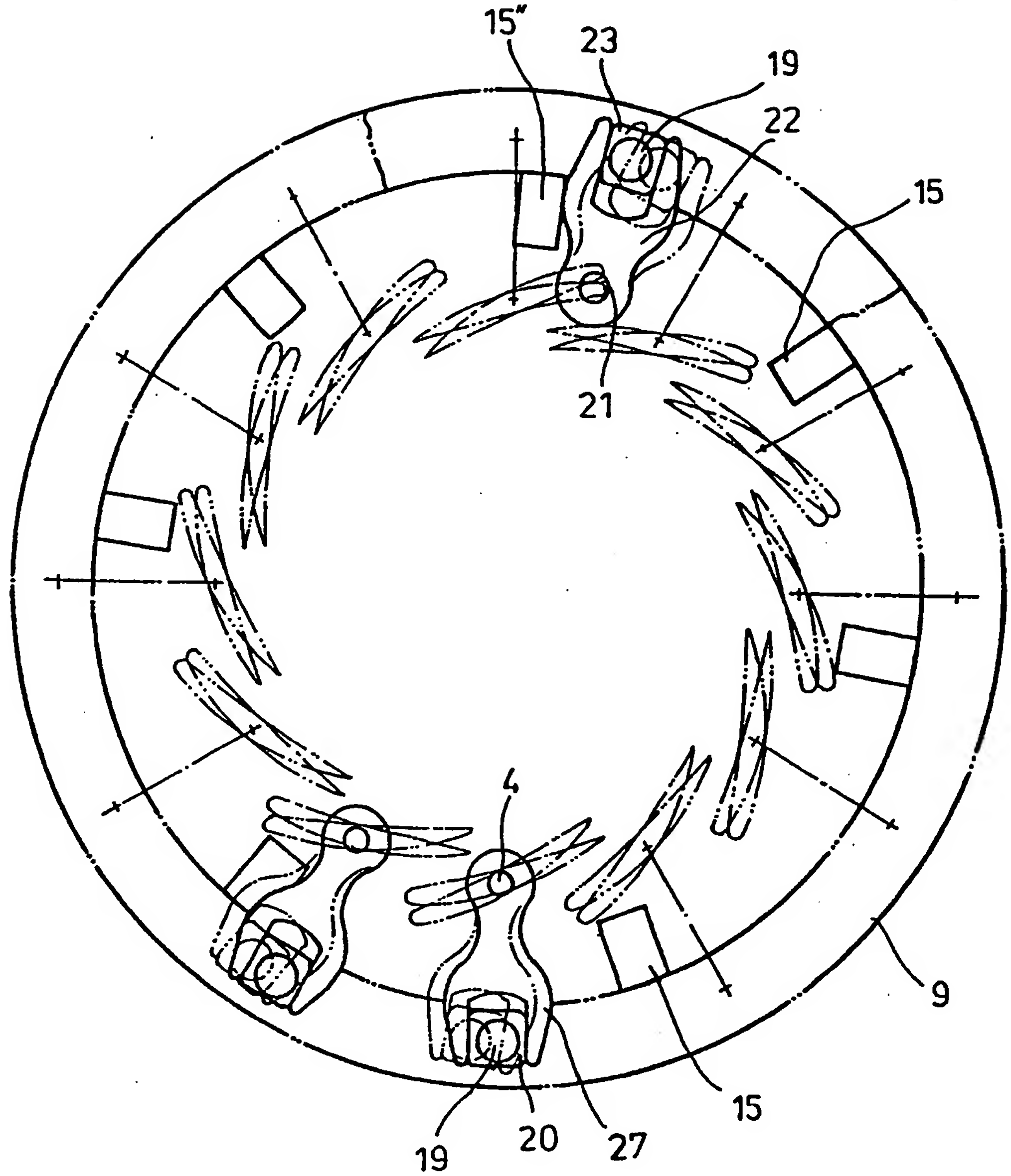

Fig. 8

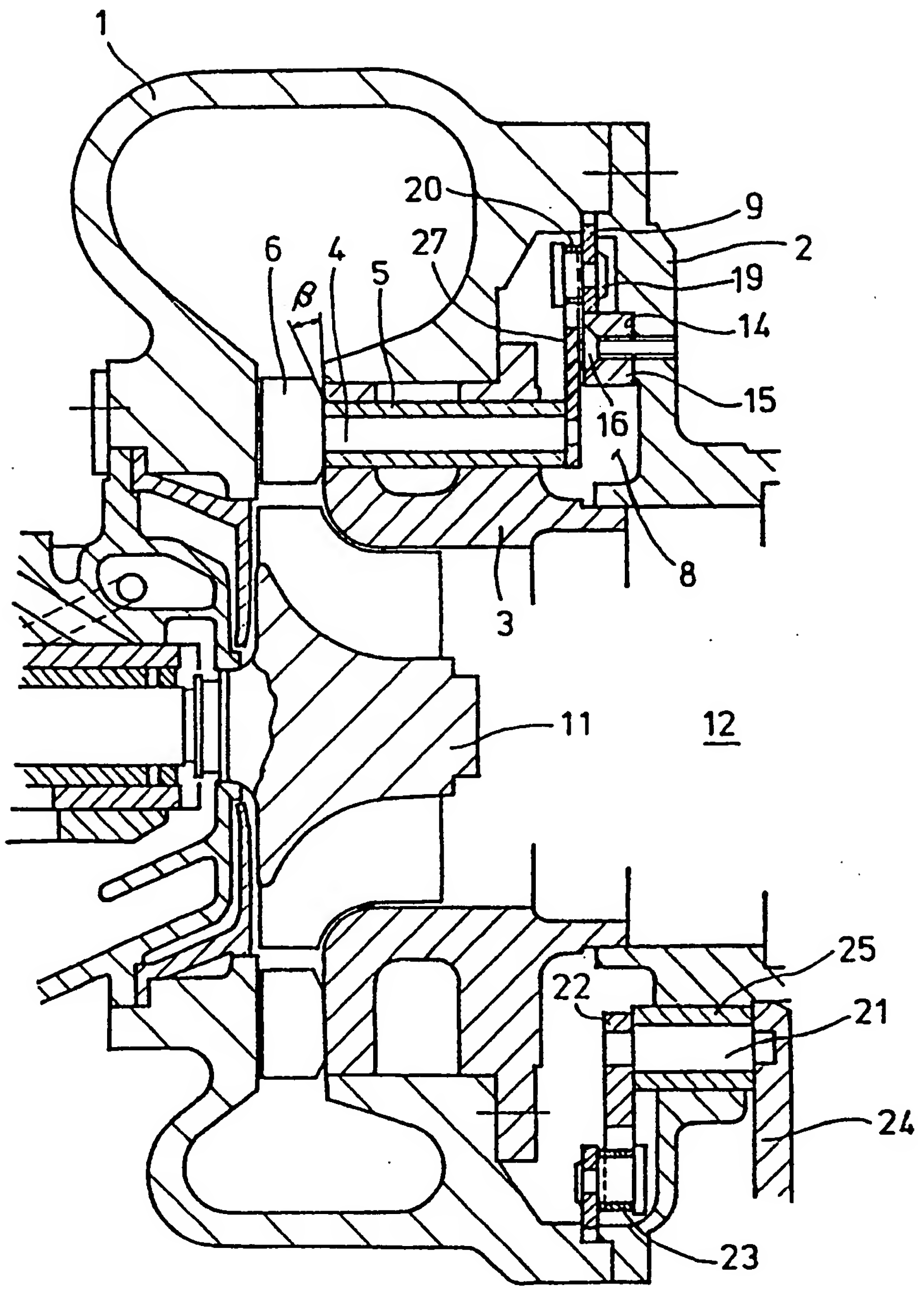

Fig. 9

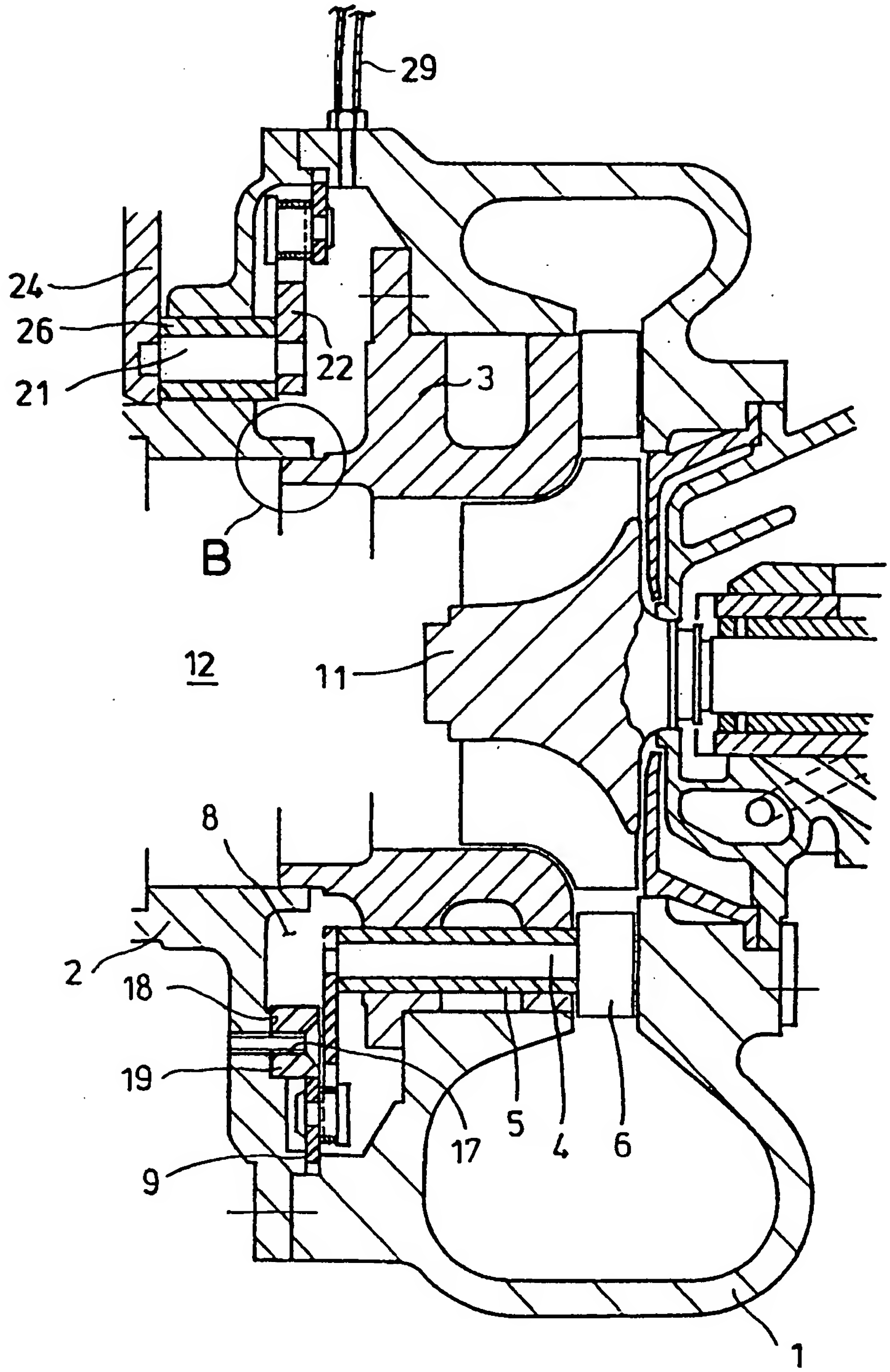

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Fig.11

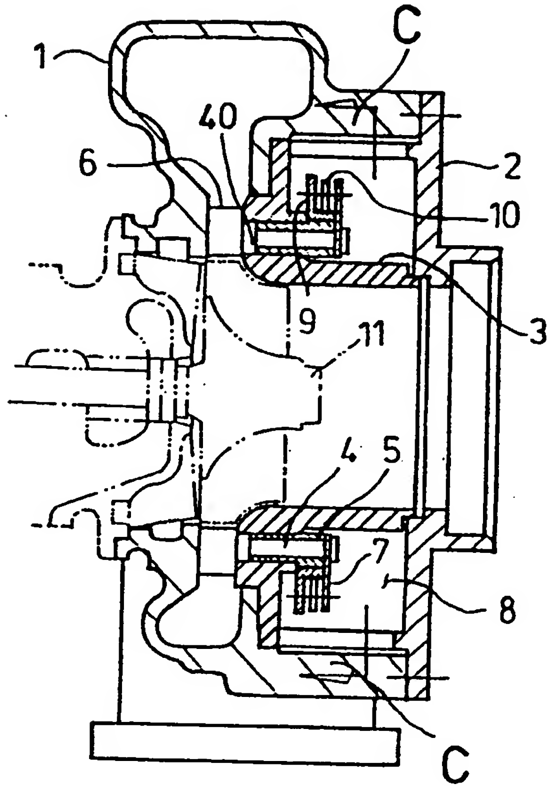

Fig.10

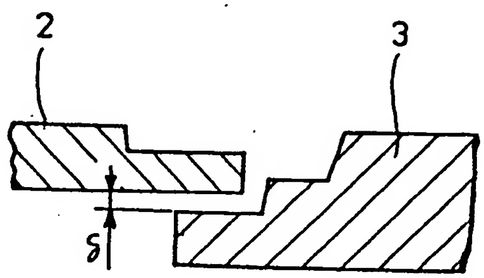

Fig.12